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Assessment of Steam-Injected Gas Turbine Systems and Their Potential Application

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SUMMARY

The performance, economics, and potential application of steam-injected gas turbine cycles were assessed. The results were arrived at by utilizing and expanding on information presented in the literature. Steam-injected gas turbines are considered for utility power generation and industrial cogeneration applications. In each application the performance and economics of steam-injected gas turbines are compared with those for simple gas turbine and combined cycles.

The efficiency and specific power of simple gas turbine cycles can be increased as much as 30 and 50 percent, respectively, by injecting steam into the combustor. However, the efficiency of a steam-injected cycle is slightly below that of a combined cycle for comparable gas turbine operating parameters.

The water consumption per unit of power of a steam-injected gas turbine falls between those of a combined cycle plant and a coal-fired steam plant when both use wet cooling towers. The cost of water treatment for steam injection would have to be much greater than the present cost of treating boiler feed-water before the water treatment cost would outweigh the cost benefit of reduced fuel consumption due to steam injection.

Economic estimates indicate that the capital investment for steam injected gas turbine systems, as well as simple gas turbine and combined cycles, is very dependent on the type of fuel. In a distillate-fuel-fired utility application a steam-injected system has a slightly higher capital cost than a simple gas turbine cycle but a significantly lower capital cost than a combined cycle. The cost of electricity favors the steam-injected system for distillate fuel costs less than \$4/GJ (~\$4/MBtu). However, there is a relatively small difference in capital cost between the steam-injected cycle and a combined cycle when both are fueled by an integrated coal gasifer. For such coal-fired cases the economic advantage of a steam-injected system over a combined cycle would be marginal.

In an industrial cogeneration application a steam-injected system could provide a wide range of power and process heat requirements and be economically competitive with both simple gas turbine and combined cycles for many applications using a clean fuel.

INTRODUCTION

The injection of steam into the combustor of a gas turbine has been known for some time to simultaneously increase the power output and system efficiency of simple-cycle gas turbines. In the past several years there has been a renewed interest in the steam-injected gas turbine concept because of an increasing economic need to conserve fuel. Steam injection increases the power output of a gas turbine by increasing the mass flow through the turbine, relative to the compressor, without significantly increasing the power required

to drive the compressor. If the steam is generated by recovering heat from the turbine exhaust, the power system efficiency is also increased. Thus, as for a conventional combined gas turbine/steam turbine cycle, the gas turbine exhaust is used to generate steam, which is then used to produce more power. However, unlike the combined cycle, the additional power is produced in the gas turbine and not in a separate steam turbine. Because the steam-injected cycle does not have a separate steam turbine or its heat rejection system, it has the potential for lower capital cost than the combined cycle.

The performance of steam-injected gas turbines has been evaluated by several authors (refs. 1 to 10). Reference 1 presents a comprehensive analysis of steam-injected gas turbines over wide ranges of temperature (815.5° to 1371° C (1500° to 2500° F)), compressor pressure ratio (6 to 24), and injected-steam-to-compressor-airflow ratio (0 to 0.40). The author reports cycle efficiency increases for steam injection of 25 to 35 percent over that for the simple cycle. The specific power of the steam-injected cycles increased continuously for increased amounts of steam injection. Reference 6 describes the performance of a steam-injected gas turbine within the limits imposed by a typical compressor surge margin, minimum stack gas temperature, and visible exhaust plume formation. The author shows that efficiency and specific power increases of 31 and 55 percent, respectively, are possible before compressor pulsations start. Reference 7 compares the performance of a steam-injected gas turbine with a combined cycle using a conventional steam bottoming cycle over a range of pressure ratios. Reference 7 also estimates qualitative economic advantages of steam injection in a utility power application. Reference 8 compares a steam-injected gas turbine to a simple-cycle gas turbine and to a combined cycle on the basis of efficiency, specific power, and the economics of producting electricity. In that reference the economic comparisons show a potential advantage for the steam-injected gas turbine over the simple-cycle gas turbine and a cost of electricity comparable to that of the combined cycle at high capacity factors. These studies considered design-point operation with clean fuel. Reference 9 describes the off-design performance of a gas turbine powerplant augmented by steam injection using low-, intermediate-, and high-heating-value coal-derived fuels.

In addition to studies, actual performance tests of a steam-injected gas turbine have been made. Test results of injecting steam into the combustor of a Westinghouse 191-G gas turbine are reported in reference 10. This machine had a design compressor pressure ratio of approximately 6.5 and produced 12 MW of electricity without steam injection. There was a 20 percent increase in generated electricity at a steam-air ratio of approximately 0.05. This gas turbine was operated continuously with steam injection for 3000 hours without any ill effects to the turbine blades.

The conclusion reached in all of these investigations is that good performance, in terms of both output and efficiency, can be achieved by injecting steam into the combustor of a gas turbine, even with state-of-the-art operating parameters.

This report presents an assessment of the performance of steam-injected gas turbines as compared with that of simple-cycle gas turbines and conventional combined cycles. Previous work on steam-injected gas turbine power systems is utilized and expanded on to identify possible areas of application where steam injection could have some competitive advantage over other power systems in terms of higher efficiency and/or lower system cost. Central-station utility power generation and industrial applications are both considered. In each application, performance and cost of the steam-injected gas turbine are compared with those of a simple-cycle gas turbine and a combined cycle.

DESCRIPTION OF SYSTEM

The basic steam-injected gas turbine cycle is shown schematically in figure 1. It is similar to a simple gas turbine cycle in that ambient air is pressurized in a compressor, heated in a combustor by the burning of fuel, and expanded in a turbine. The turbine drives the compressor and generator. The heat in the turbine exhaust (otherwise rejected in a simple cycle) is used to raise relatively low-pressure steam in a heat-recovery boiler. This steam is injected into the combustor, where it is heated to the turbine-inlet temperature.

Steam injection provides an additional control parameter and hence allows an increase in the operational flexibility of the system over that for the simple cycle. The net power output can be changed by changing the amount of steam injected into the combustor and by simultaneously changing the fuel input rate to maintain a constant turbine-inlet temperature. There are several possible limitations on the maximum amount of steam that can be injected. These limitations are discussed in the following section. In addition, if there is some other on-site requirement for process steam, the steam generated in the heat-recovery heat exchanger can be divided between that for injection and that for process.

THERMODYNAMIC PERFORMANCE

A map of the performance, at possible design points, of a steam-injected gas turbine cycle is shown in figure 2 for ranges of injected-steam-to-compressor-airflow ratios (S/A) and heat-recovery steam generator (HRSG) design-point parameters. These results were calculated by using the following cycle parameters:

Turbine-inlet temperature, °C (°F)	1093.3° C (2000° F)
Compressor pressure ratio	16
Compressor and turbine polytropic efficiency	0.87
Compressor inlet temperature, °C (°F)	15 (59)
Loss pressure ratio	0.92
Liquid fuel	Conventional
Air cooling in the turbine	Yes

It was assumed that the steam was generated at a pressure equal to that at compressor discharge. The heat-recovery steam generator approach temperature difference ΔT_{ap} (the temperature difference between the exit steam flow and the inlet gas flow) and the pinch-point temperature difference ΔT_{pp} (the minimum temperature difference between the hot- and cold-side fluids at any point within the HRSG) were parametrically varied. As the design steam injection rate was increased relative to the compressor airflow (increasing S/A), more power was produced in the turbine relative to the power required by the compressor, resulting in the increase in specific power shown in the figure. Furthermore, as the steam injection rate was increased, more heat was recovered from the exhaust and returned to the cycle. Along a line of constant ΔT_{ap}, as the steam injection rate was increased, the increased exhaust heat recovery had a greater effect than the increased amount of fuel needed to heat the steam to the turbine-inlet temperature, and therefore both cycle efficiency and specific power increased. Along a line of constant ΔT_{pp}, an increase in S/A resulted in a lowering of the steam temperature at the exit of the HRSG (increase in ΔT_{ap}). Additional fuel would be required

to heat this lower temperature steam to the turbine-inlet temperature. Thus, despite the further increase in the amount of exhaust heat recovery, the additional fuel input outweighed the incremental gain in specific power because of the increased steam flow, and cycle efficiency was reduced. This trend continued until the stoichiometric fuel-air ratio (zero excess air) was reached in the combustor. Any further increase in S/A would result in a reduction in the turbine-inlet temperature.

The choice of design values for ΔT_{ap} and ΔT_{pp} is limited to some minimum value for a practical HRSG. Generally a value of 10° C (50° F) is considered reasonable for ΔT_{ap} and ΔT_{pp}. At this value a cycle efficiency increase of approximately 30 percent, as compared with the simple-cycle gas turbine, occurs at a steam-air ratio of approximately 0.16. The increase in specific power at this point is about 65 percent.

There are several other limitations on system design that are illustrated in figure 2. The gas temperature at the HRSG exit is generally constrained to remain above the dewpoint of any corrosive constituents to insure that these constituents do not condense in the HRSG or the exhaust stack. Design values for stack inlet temperatures are typically in the range 121° to 149° C (250° to 300° F) when a margin is included above the acid dewpoint. If 149° C (300° F) is used as the minimum allowable exhaust gas temperature, the minimum limitation occurs before the water dewpoint is reached in the HRSG and also occurs very near the value of S/A where a ΔT_{ap} and ΔT_{pp} of 10° C (50° F) are reached. If a clean fuel with very low sulfur content is used, the minimum allowable stack inlet temperature to avoid acid condensation might be lower than 149° C (300° F).

As a result of steam injection the partial pressure of the water vapor in the exhaust gas is relatively high. The locus of points where the water vapor in the exhaust gas will condense (dewpoint) in the exhaust side of the HRSG is also indicated in figure 2. Another possible concern in some applications might be the formation of a visible condensation plume in the atmosphere. A visible plume could form in the atmosphere at steam-air ratios considerably less than that which would cause water vapor to condense in the exhaust side of the HRSG.

Still another possible limitation on the steam-air ratio is the point where the steam produced in the HRSG is saturated. (It might be desirable to maintain a minimum degree of superheat in the injected steam.) The locus of points where wet steam starts to leave the HRSG is also indicated in figure 2. The discontinuity along lines of constant ΔT_{pp} at an S/A of approximately 0.25 corresponds to a change in the number of turbine stages and hence to a discontinuous change in the turbine cooling requirements. For other gas turbine conditions the limits shown in figure 2 would be expected to shift relative to each other. Similar trends are reported in references 1 and 4 to 8 for various gas turbine operating parameters. All show significant performance improvements with steam injection when compared with a simple-cycle configuration.

Also shown in figure 2 is the performance of a combined gas turbine/steam turbine cycle. The combined-cycle performance was based on the same gas turbine operating parameters that were used in the steam-injected cycles. The steam turbine throttle pressure was assumed to be 22.8 atm (335 psia), a suitable pressure to be used in connection with the 491° C (916° F) exhaust gas temperature of the gas turbine. In a single-pressure HRSG, the lower the steam pressure, the more of the heat in the gas turbine exhaust that can be used to generate steam. Throttle pressures that are too low reduce steam turbine performance. The throttle pressure selected was a compromise that

tended to maximize the combined-cycle performance. HRSG approach and pinch-point temperature differences ΔT_{ap} and ΔT_{pp} of 10° C (50° F) were used for this calculation. The selection of a ΔT_{ap} is equivalent to selecting a steam turbine throttle temperature. The steam condenser pressure was 0.079 atm (1.16 psia). Two regenerative feedwater heaters were used, one of which was a deaerator. The cycle efficiency of the combined cycle was 41.3 percent as compared with 39.0 percent for the steam-injected cycle at ΔT_{ap} and ΔT_{pp} of 10° C (50° F). However, at these conditions, the specific power based on compressor airflow for the steam-injected cycle was nearly 23 percent greater than that of the combined cycle. Similar results for comparisons between steam-injected gas turbines and combined cycles are reported in references 4, 7, and 8. References 4 and 7 report that the combined cycle had 4 percentage points higher efficiency than the steam-injected cycle for a gas turbine inlet temperature of 1427° C (2600° F) and a compressor pressure ratio of 16. For this condition the steam-injected cycle had 42 percent greater specific power than the combined cycle on the basis of compressor airflow. The results reported in reference 8 show the combined cycle with a 3 percentage point higher efficiency than the steam-injected cycle and the steam-injected cycle with a 10 percent higher specific power. Both of these cycles had a 1204° C (2200° F) turbine-inlet temperature. But the combined cycle had a compressor pressure ratio of 12 as compared with 16 for the steam-injected cycle. The lower pressure ratio for the combined cycle allowed the use of higher steam-cycle throttle conditions than was considered in developing figure 2.

Information presented in reference 3 indicates that the full potential of steam injection is achieved at rather high compressor pressure ratios of 20 to 30 for a turbine-inlet temperature of 1093° C (2000° F). The use of these higher pressure ratios in a steam-injected gas turbine would alter the comparison with the combined cycle, which has a gas turbine pressure ratio of 12 to 16 for optimum efficiency. The information presented in reference 3 apparently does not include the effect of turbine cooling. Reference 1 does show the effect of turbine cooling on the performance of steam-injected gas turbines. This report also indicates that pressure ratios in excess of 20 are required for maximum efficiency of an uncooled 1093° C (2000° F) turbine. However, for an air-cooled turbine, the pressure ratio for maximum efficiency is reduced to approximately 16. For a cooled turbine (at a fixed turbine-inlet temperature) more turbine stages are required as the pressure ratio is increased, and thus more compressor airflow is required for blade and vane cooling and blockage flow. This increased coolant flow reduces the work-producing flow through the turbine and thus reduces system efficiency as compared with the cycle using an uncooled turbine.

The actual consumption of water in a steam-injected gas turbine does not appear to be a concern. However, the cost associated with treating it may be. Based on the information presented in figure 2, the water consumption would fall between 1.04 and 1.63 kg/kW-hr (2.3 and 3.6 lb/kW-hr) depending on the particular pinch-point and approach temperature differences used. The water consumption falls between the 0.77 and 1.72 kg/kW-hr (1.69 and 3.8 lb/kW-hr) quoted in reference 8 for a combined-cycle plant and a coal-fired steam plant when both use wet cooling towers. Reference 7 indicates that the cost of treated water would have to reach between \$4 and \$11 per kiloliter (\$15 and \$40 per 1000 gallons) before the water and fuel costs of a steam-injected cycle exceed the fuel cost of a simple-cycle gas turbine. Thus the cost of treated water could be considerably above the present \$0.3 to \$0.5 per kiloliter (\$1 to \$2 per 1000 gallons) for boiler feedwater treatment.

The thermodynamic results discussed thus far apply to design-point operation, where the turbomachinery would be designed for the flow mismatch between the turbine and the compressor for each value of steam-air ratio. A potentially attractive feature of the steam-injected gas turbine cycle would be its use to augment the power output of existing simple cycles. In fixed-geometry machines not specifically designed for steam injection the relative increase in mass flow through the turbine with steam injection results in an increase in compressor backpressure and therefore an increase in the compressor pressure ratio. As the steam injection rate increases, the compressor pressure ratio increases toward its surge limit. The maximum increase in performance obtainable with steam injection then depends on the surge margin available to the particular compressor. This concept was demonstrated to a limited extent in reference 10. Tests were conducted on a Westinghouse 191-G gas turbine. No special modifications were done to the machine except to incorporate a steam line and install new standard baskets in the combustor. The 191-G had a compressor pressure ratio of approximately 6.5 and produced 12 MW of electricity without steam injection. At a steam injection rate of 15 876 kg/hr (35 000 lb/hr), a steam-air ratio of less than 0.05, the generated output increased by approximately 20 percent. At this steam flow the compressor pressure ratio increased to 6.9. Although this steam-air ratio is lower than the injection rates shown to be desirable in figure 2, it did produce a significant increase in power. The use of larger steam injection rates would probably have required major modifications to the combustor.

The off-design performance of a specific gas turbine powerplant augmented by steam injection was estimated in reference 9. In this report the results of using coal-derived low- and intermediate-heating-value fuel gas are compared with the results of using a conventional distillate. The turbomachinery characteristics used in this analysis were scaled from available performance maps of aircraft gas turbines. Calculations were performed with the assumption that the turbomachiery operated at its design point for the low- and intermediate-heating-value fuel gas without steam injection. Operating points were then determined for steam-injection rates and for each of the other fuels with and without steam injection. The results indicate that steam injection could provide substantial increases in both power (68 to 100 percent) and efficiency (32 to 52 percent) within the surge margin of that particular compressor for all fuels.

Based on the calculations presented in this section and in the literature, steam injection is an attractive option for increasing both the efficiency and specific power of simple-cycle gas turbines. Although the efficiency of steam-injected cycles is lower than that of combined gas turbine/steam turbine systems, the specific power is higher.

CAPITAL COST

It has been shown in the previous section that, although steam injection does significantly increase the efficiency of simple-cycle gas turbines, it still falls somewhat below that of combined gas turbine/steam turbine systems. The potential attractiveness of steam-injected gas turbines also depends heavily on how their costs compare with those of competing systems. One would expect a steam-injected system to have a higher capital cost than a simple-cycle gas turbine because of the additional costs of the heat-recovery heat exchanger and the water supply and treatment system. But compared with combined-cycle systems the capital cost of the steam-injected gas turbine should be lower because there is not a separate steam turbine and its associated heat-rejection system.

Only a few authors have considered cost in the evaluation of steam-injected cycles. Two references (8 and 11) have included estimates of capital cost in their evaluation. The capital cost estimates of reference 8 are summarized in table I. The economic data from ECAS Phase I (ref. 12) were used, wherever possible, as an existing data base to estimate the equipment capital cost for this study. In the costing analysis, it was assumed that all systems were fired by a distillate fuel. The costs are expressed in 1974 dollars to be consistent with ECAS Phase I results. A contingency of 20 percent was added to the installed cost to cover items not detailed when considering only major components. An adder of 11.8 percent for a 1-year construction period and 20.4 percent for a 2-year construction period was used to account for interest and escalation costs during the construction. The estimates shown in table I apply to utility-size machines and indicate that the total capital cost of the steam-injected cycle is about 3 percent more than that of the simple cycle and about 33 percent less than that of the combined cycle.

Similar capital cost differences were estimated by the United Technology Corp. for the comparisons made in reference 11. The information presented and discussed thus far has shown that the steam-injected gas turbine system has higher efficiency and slightly higher capital cost than a simple cycle. As compared with a combined cycle, the steam-injected system has slightly lower efficiency but considerably lower capital cost. To evaluate the competitiveness of the steam-injected system, the overall cost of ownership and operation must be considered. This is the subject of the next two sections, where steam-injected systems are evaluated separately for both utility and industrial applications. The utility and industrial applications are treated separately because the plant size, economic criteria, and appropriate fuel for each application are different and could alter the comparisons between competing systems.

EVALUATION OF STEAM INJECTION FOR UTILITY APPLICATION

Central-station utility application as used in this discussion is limited to large power systems producing only electricity. For this application, steam-injected cycles are compared with conventional steam and combined cycles for base and intermediate loads and with simple-cycle gas turbines for peaking operation.

The cost of electricity (COE) for a simple cycle, a steam-injected cycle, and a combined cycle were evaluated and compared in reference 8 for a wide range of capacity factors. The COE is composed of capital cost, operation and maintenance (O&M) cost, and fuel and water costs. The capital component was calculated by using a fixed charge rate of 18 percent of the total plant capital cost. The O&M component was 2 mills/kW-hr. The distillate oil price was \$2.84/GJ (\$3.00/MBtu). Water and water treatment costs were based on water consumption. As compared with the simple cycle, the steam-injected cycle has a lower COE except at very low capacity factors (less than 0.015). As compared with the combined cycle, the steam-injected cycle has marginally lower COE for high-to-intermediate capacity factors (1.0 to 0.5) and significantly lower COE for low capacity factors.

As mentioned previously, the fuel contribution to COE in reference 8 was based on a distillate oil price of \$2.84/GJ (\$3.00/MBtu). At present the cost of residual oil delivered to a utility is approaching \$4.74/GJ (\$5.00/MBtu). The effect of fuel price (oil or natural gas) on the COE is shown in figure 3 over a range of plant capacity factors. The information presented in this

figure was calculated by using the same capital, O&M, and water costs established in reference 8 but for a fuel price ranging from \$2.84/GJ to \$9.48/GJ (\$3.00/MBtu to \$10.00/MBtu). As one would expect, the cost advantage of the steam-injected cycle over the simple cycle at lower capacity factors increases as the fuel price increases. At the present fuel price, one would not anticipate large energy savings on a national basis from peaking application (i.e., low capacity factor). However the use of steam injection for peaking could become more economically attractive as fuel prices are escalated. The marginal cost advantage of the steam-injected cycle, as compared with the combined cycle reported in reference 8, at high-to-intermediate capacity factors is reversed in favor of the combined cycle for fuel prices exceeding \$3.8/GJ to \$4.7/GJ (\$4/MBtu to \$5/MBtu).

The comparisons discussed thus far are applicable to power systems burning clean fuel. The Fuel Use Act of 1978 requires that new electric generating units consider burning coal as the primary fuel. In light of this requirement, it is of interest to evaluate the economics of steam-injected gas turbines with integrated gasifiers for utility application. Table II presents a capital cost comparison of a steam-injected gas turbine system and a combined cycle, both using an integrated gasifer as the fuel source. The capital costs of the steam-injected gas turbine and the gas and steam turbines of the combined cycle were taken from reference 8 and escalated to 1975 dollars (escalation rate of 6.5 percent). The capital costs of the gasifier and the balance-of-plant equipment, the contingency cost, and the costs of escalation and interest during construction were taken from reference 13. The specific cost of a gasifier producing low-heating-value fuel gas was \$66.0/kW of coal. The specific cost in dollars per kilowatt of electricity was obtained by dividing the cost in terms of coal by the product of powerplant efficiency (using clean fuel) and the gasifer/cleanup system efficiency (86.2 percent). The balance-of-plant cost was \$40.3/kW of coal for the steam-injected cycle and \$43.2/kW of coal for the combined cycle. The difference in the balance-of-plant cost is the absence of the cooling tower requirements in a steam-injected cycle. Architect and engineering services were assumed to be 10 percent and contingency, 20 percent. A construction period of 5 years was assumed for both the combined-cycle and steam-injected systems. The capital costs were increased by 1.487 for escalation and interest during the 5-year construction period. It was assumed in this comparison that the gasifier was the pacing item in construction and that the absence of a steam turbine and its heat-rejection system in the steam-injected gas turbine power system would not significantly reduce the construction period in comparison with the combined cycle. As can be seen in table II there is little difference between the total capital costs of the steam-injected gas turbine and combined-cycle systems when both use fuel from a gasifier. The savings in the steam-injected system due to the absence of the steam turbine and heat-rejection component costs in a clean-fuel-fired system are masked by the gasifier and balance-of-plant costs in a gasifier-fueled system. However, in this case, the combined cycle would have a lower fuel cost leverage on COE because of the lower price of coal as compared with a clean fuel. Therefore the total costs of electricity for the two systems would be very nearly the same.

EVALUATION OF STEAM INJECTION FOR INDUSTRIAL APPLICATIONS

Most industrial applications require only a small fraction of the power-generating capacity of a typical utility powerplant. However, few industries can generate electricity as economically as utilities because they cannot take

advantage of economies of scale. But there are many industrial applications where both heat and electricity could be generated (cogeneration) on site at higher overall efficiency than when power and heat are generated separately. Few central-station utility powerplants can take advantage of the potential economies of cogeneration. A utility can benefit from cogeneration only if there are industrial customers close by who can purchase large quantities of steam on a continuous basis.

Comparisons of steam-injected gas turbines with either simple cycles or combined cycles for an industrial application would have to consider the influence of unit size on the performance and cost and their relative abilities to provide both power and process steam for a range of site-required power-heat ratios. Both steam-injected gas turbines and combined cycles can provide a range of power-to-process-steam ratios in a cogeneration application.

In a steam-injected cycle, part of the steam generated in the heat-recovery steam generator can be injected into the gas turbine combustor, and part can be used for process. By varying the relative amounts, the power-heat ratio can be varied. The degree of flexibility of steam injection in a cogeneration application is illustrated in figure 4. This figure, taken from reference 9, illustrates the potential for these systems to follow variations in power and process steam requirements. Three of the four gas turbine cycles shown were integrated with gasifiers; the fourth used a distillate fuel. In the cycles with the integrated gasifiers, the process steam includes steam generated in the gasifiers. The power output from these systems can vary as much as 145 percent, as the process steam requirements go from maximum to zero, without exceeding the compressor surge margin.

In a combined cycle, if an extraction steam turbine is used, the extraction rate can also be varied to match a range of process steam requirements. The effect of varying the steam extraction rate on the efficiency and power-heat ratio of a combined cycle is shown in figure 5. Also shown is the effect of the steam injection rate (steam-air ratio) on the system efficiency and power-heat ratio in a steam-injected gas turbine. The system parameters used to calculate the performance of these two cycles are shown in table III. The power-heat ratio for the combined cycle can be varied from infinity at an extraction rate of zero to 1.2 at an extraction rate of 85 percent of throttle flow. The combined-cycle efficiency decreases linearly for increasing extraction rates. Also shown in figure 5 is the effect of steam turbine efficiency on the performance of the combined cycle. One concern with combined cycles is that for small powerplants the steam turbine is relatively small and the expansion efficiency is relatively low. The efficiency range of commercially available steam turbines is shown in reference 14 for power levels of a few kilowatts to 100 MW. For a power level of 100 MW the efficiency is shown to range from percentages in the high 70's to low 80's. The efficiency of the more efficient 100-kW steam turbines are in the 60 percent range. The range of steam turbine efficiencies shown in figure 5 could apply to combined-cycle power levels of 300 kW to more than 300 MW. A decrease in steam turbine efficiency from 80 to 60 percent decreases the combined-cycle efficiency by 2 to 4 percentage points depending on the steam extraction rate. Because the efficiency of gas turbines is in general much less dependent on size effects, the performance of steam-injected gas turbines could approach that of low-power-level combined cycles for higher power-heat ratios. The performance of a simple-cycle gas turbine is indicated by the steam-air ratio of zero on the steam-injected gas turbine curve.

In a cogeneration application a more appropriate measure of performance is the fuel-energy-saving ratio (FESR). The fuel-energy-saving ratio is defined as

$$\mathrm{FESR} = \frac{Q(\mathrm{noncogeneration}) - Q(\mathrm{cogeneration})}{Q(\mathrm{noncogeneration})}$$

where the Q's are the respective fuel energy requirements for the noncogeneration and cogeneration situations. Figure 6 shows the cogeneration fuel energy savings corresponding to the curves shown in figure 5. The noncogeneration fuel energy requirement is obtained by assuming that the required power is generated by a 32-percent-efficient steam plant and that the required steam is generated by a 85-percent-efficient boiler so that

$$Q(\mathrm{noncogeneration}) = \frac{(\mathrm{Power\ required})}{0.32} + \frac{(\mathrm{Steam\ required})}{0.85}$$

The fuel energy savings for all three systems, the simple cycle ($S/A = 0$), the steam-injected cycle ($S/A > 0$), and the combined cycle, range between 28 and 39 percent. The combined cycle, with an extraction turbine, yields the highest fuel energy saving over the range of power-heat ratios shown. Basically this is because power is produced by all of the steam before a portion of it is used for process. However, the advantage enjoyed by high-power-level combined cycles (i.e., steam turbine efficiency of ~80 percent) is significantly reduced in low-power-level applications (60 percent steam turbine efficiencies) and may even disappear for high power-heat applications. This, together with the lower capital investment of steam-injected systems, could make them more attractive than combined cycles in high power-heat applications.

The economics of combined cycles and steam-injected cycles in cogeneration applications was evaluated in references 11 and 14. These two reports are the result of the Cogeneration Technology Alternatives Study (CTAS). These studies were aimed at providing a data base to assist the U.S. Department of Energy in deciding R&D funding priorities in the area of energy conversion system technology for advanced industrial cogeneration applications.

They matched over 50 industrial processes with conversion systems to evaluate the energy saving, environmental impact, and economic viability of cogeneration as compared with the traditional methods of providing the industrial process energy requirements. The industries considered had electricity requirements ranging from 320 kW to 200 MW and power-heat ratios ranging from 0.002 to 2.17. These energy conversion systems were either sized to meet the process heat requirements (in which case electricity is either bought or sold) or sized to meet the electrical requirements, in which case an auxiliary boiler is usually required to supply any heat needs.

Steam-injected gas turbines were among the energy conversion systems considered. In reference 11, two design options for the steam-injected cycles were selected at a turbine-inlet temperature of 2500° F and a pressure ratio of 18: one with a steam-air ratio of 0.05; and the other with a steam-air ratio of 0.10. Two of the main parameters used in CTAS to compare the energy-conversion systems were the fuel-energy-saving ratio and the cost-saving ratio. The FESR is as previously defined. The CSR is defined as

$$CSR = \frac{LAC(\text{noncogeneration}) - LAC(\text{cogeneration})}{LAC(\text{noncogeneration})}$$

where LAC's are the respective levelized annual costs. The levelized annual cost is that constant level cost that, when distributed over the lifetime of a system, would have the same present value as the actual stream of costs when both are discounted at the cost of capital.

The results presented in reference 11 show a significant variation in the FESR and CSR for each energy conversion system from one cogeneration application to another. The best cogeneration application for the combined cycle had an FESR of approximately 0.41 when the system was sized to meet the electrical requirements. The best cogeneration applications for the simple-cycle gas turbine and the steam-injected gas turbine both had an FESR of near 0.38. The range of FESR values for all applications was about the same for the combined cycle, the simple cycle, and the steam-injected cycle. When the conversion systems were sized to meet the process heat requirement, the combined cycle and simple cycle had a best-application FESR value of 0.38 and the steam-injected cycle had a value of 0.31.

The maximum cost-saving ratio (CSR) for each of these three systems was 0.28. The range of CSR values for all applications was also the same for each of the three systems. The combined cycle, the simple cycle, and the steam-injected cycle each produced a positive annual cost saving in nearly 80 percent of the cogeneration applications considered.

CONCLUDING REMARKS

Both the net power output and efficiency of gas turbine systems can be increased by injecting steam into the combustor of the gas turbine. Steam injection increases the power output by increasing the mass flow through the gas turbine without significantly increasing the power required to drive the compressor. If the injected steam is generated by recovering otherwise wasted heat from the turbine exhaust, the system efficiency is also increased.

Numerous investigators have evaluated the performance of steam injection systems over a wide range of operating parameters. Some have compared their results with those obtained for simple and combined-cycle power systems. The thermodynamic results presented herein and cited from the literature substantiate that steam injection is an attractive means of increasing both the power and efficiency of gas turbines. Efficiency increases of 20 to 30 percent can accompany specific power increases of 30 to 50 percent by the use of steam injection in simple gas turbines. The degree of increase in performance was shown to depend on the steam injection rate (steam-air ratio).

The maximum steam-air ratio that can be used in a particular system is in turn established by system design considerations (i.e., turbine-inlet temperature, compressor pressure ratio, and HRSG approach and pinch-point temperature differences). The efficiency of steam-injected cycles, however, is slightly below that of combined cycles for comparable gas turbine operating parameters.

The attractiveness of steam-injected gas turbines also depends heavily on how they compare in cost with competing systems for various applications. There is not a great deal of information available in the literature on cost comparisons involving steam-injected cycles. What information there is applies mostly to large utility power systems burning distillate fuel. These data indicate that steam-injected cycles have a capital cost considerably

lower than combined cycles but slightly higher than simple cycles. As a result, the steam-injected cycle in a utility application would have a lower cost of electricity (COE) than a simple cycle for all but very low capacity factors. A steam-injected cycle using distillate-grade fuel at $2.84/GJ ($3.00/MBtu) has marginally lower COE for high-to-intermediate capacity factors (1.0 to 0.5) and significantly lower COE for low capacity factors than a combined cycle.

The COE advantage of a steam-injected cycle over a simple cycle increases as fuel prices escalate. However, the marginal cost advantage of the steam-injected cycle, as compared with a combined cycle at high-to-intermediate capacity factors, is reversed in favor of the combined cycle for fuel prices exceeding $3.8/GJ to $4.7/GJ ($4/MBtu to $5/MBtu).

Economic estimates presented in this report indicate a relatively small difference in capital cost between a steam-injected cycle and a combined cycle when both systems use a fuel derived from an integrated coal gasifier. The capital cost of the power-producing portion of a steam-injected cycle is less than that for the combined cycle. This lower cost is masked by the addition of the large cost associated with a gasifier system in a coal-derived fuel case. However, when using a gasifier, the combined cycle would have a lower fuel cost leverage on COE because the cost is lower for coal than for a clean fuel. Therefore the total COE's for the steam-injected and combined cycles would be very nearly equal.

Both steam-injected and combined cycles can satisfy a range of power and process heat requirements for industrial cogeneration applications. In a steam-injected cycle, part of the steam generated in the heat-recovery steam generator can be injected into the gas turbine combustor to increase power during periods when that steam is not required for process. The maximum increase in power that can be achieved by this steam injection depends on the surge margin of the particular compressor. In the combined cycle, the power output can be increased by decreasing the amount of steam extracted for process requirements.

System power level has a considerably larger effect on the performance of a combined cycle than it does on the performance of a steam-injected cycle. This is because the expansion efficiency of commercially available steam turbines decreases for decreasing power levels. The efficiency of gas turbines is, in general, much less dependent on power level. As a result the performance of the steam-injected gas turbine, which is normally less than that of the combined cycle, could approach that of the combined cycle for low-power-level, high-power-heat-ratio applications. These results together with the lower capital investment of the steam-injected cycle could make it more attractive than the combined cycle for cogeneration applications requiring small power levels and using a clean fuel.

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TABLE I. - CAPITAL COST DISTRIBUTION

[Mid-1974 dollars.]

	Simple gas turbine cycle	Steam-injected gas turbine cycle	Combined cycle
Net powerplant output, MW	80.6	127.4	475.8
Installed cost (major components), $/kW	109.2	113.0	156.2
Contingency cost, $/kW	21.8	22.8	31.3
Escalation cost, $/kW	8.7	8.6	21.0
Interest during construction, $/kW	6.7	6.3	17.2
Total capital cost, $/kW	146.4	150.7	225.7
Construction time, yr	1	1	2
Net efficiency, percent	32.4	40.5	42.5

TABLE II. - COMPARISON OF CAPITAL COST FOR A STEAM-INJECTED GAS TURBINE CYCLE AND A GAS TURBINE/STEAM TURBINE COMBINED CYCLE BOTH WITH A LOW-HEATING-VALUE GASIFIER

[1975 DOLLARS.]

	Steam-injected gas turbine cycle	Combined cycle
	Capital cost, $/kW	
Topping cycle	66.9	56.4
Heat-recovery steam generator	19.9	----
Bottoming cycle	----	39.2
Gasifier	189.1	180.2
Balance of plant	101.8	109.0
Subtotal	377.7	385.3
Architect and engineering services (at 10 percent)	37.8	38.5
Contingency (at 20 percent)	83.1	84.8
Escalation and interest	242.8	247.7
Total	741.4	756.3

TABLE III. - PARAMETERS USED IN PERFORMANCE CALCULATIONS

Parameter	Combined cycle	Steam-injected gas turbine
Gas turbine cycle:		
Compressor-inlet temperature, °C (°F)	15.6 (60)	15.6 (60)
Compressor pressure ratio	11.0	11.0
Compressor efficiency	0.846	0.846
Turbine-inlet temperature, °C (°F)	1093 (2000)	1093 (2000)
Turbine pressure ratio	9.735	9.735
Turbine efficiency	0.91	0.91
Loss pressure ratio	0.885	0.885
Steam injection ratio (m_s/m_a)	0	0,0.05,0.10,0.15
Steam turbine cycle (extraction):		
Turbine-inlet temperature, °C (°F)	538 (1000)	-----
Turbine-inlet pressure, atm (psia)	98.6 (1450)	-----
Turbine efficiency	0.60, 0.70, 0.80	-----
Two feedwater heater pressures, atm (psia)	1.4 (20.8); 0.4 (5.5)	-----
Pump efficiency	0.70	-----
Extraction rate, percent	0,10,20,30,40,50,60,70	-----
Extraction pressure, atm (psia)	4.4 (65)	-----

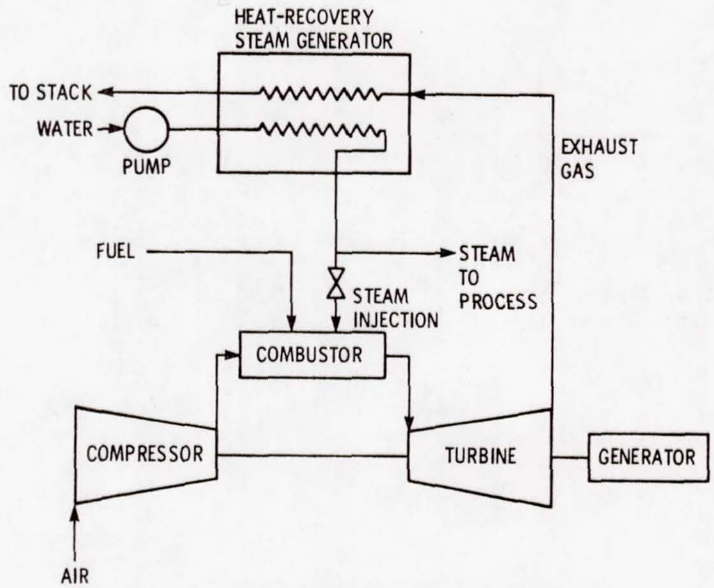

Figure 1. - Steam-injected gas turbine.

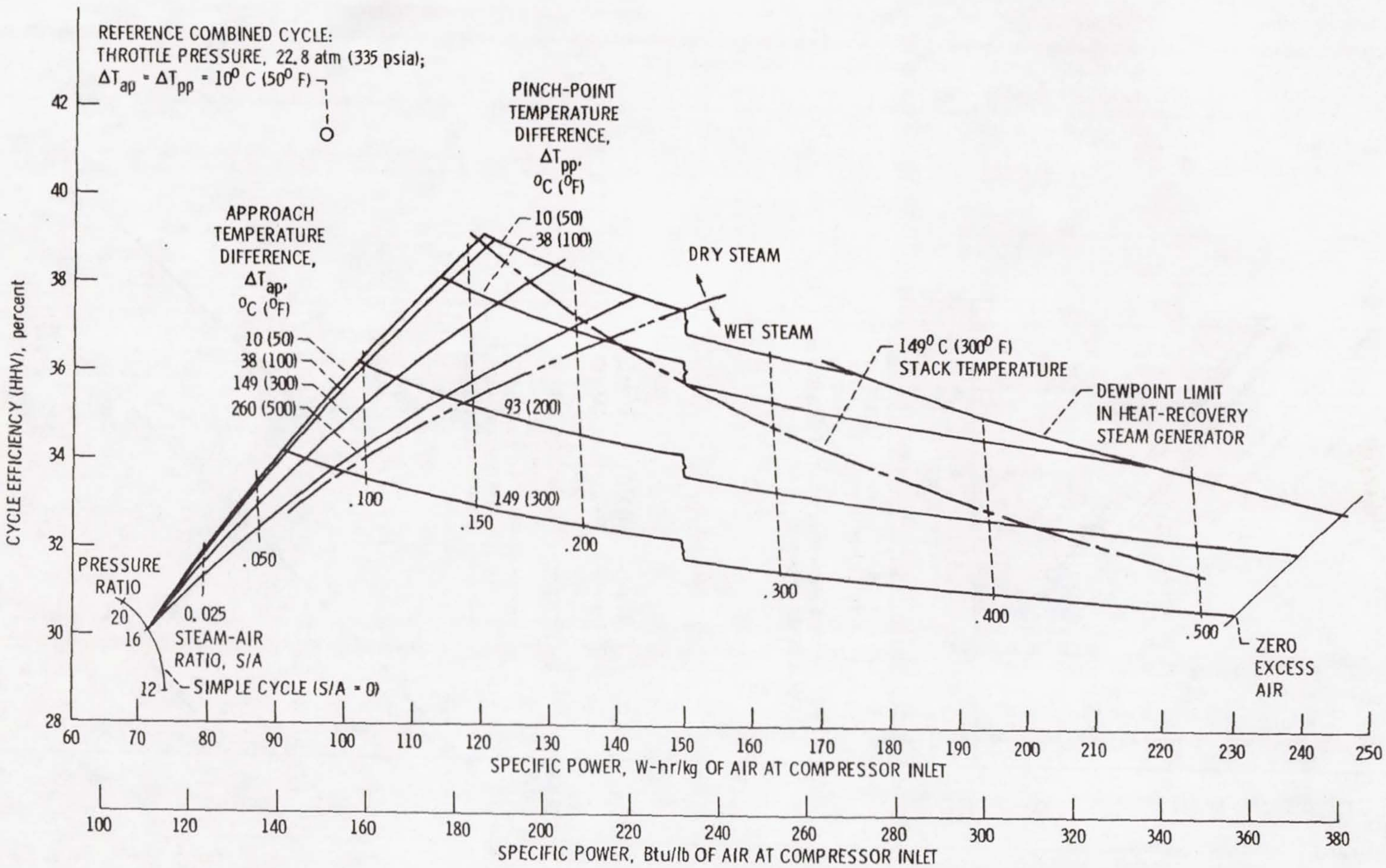

Figure 2. - Performance of steam-injected gas turbine. Pressure ratio, 16; turbine-inlet temperature, 1093^0 C (2000^0 F); air cooled.

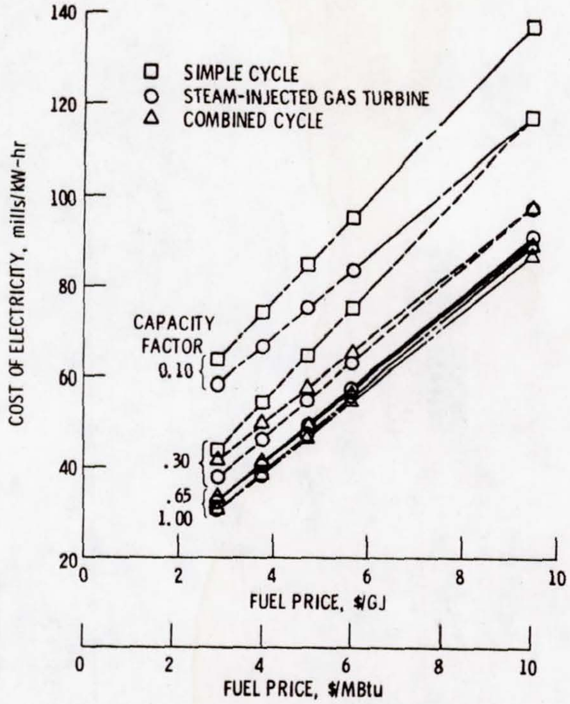

Figure 3. - Effect of fuel price on cost of electricity over a range of capacity factors.

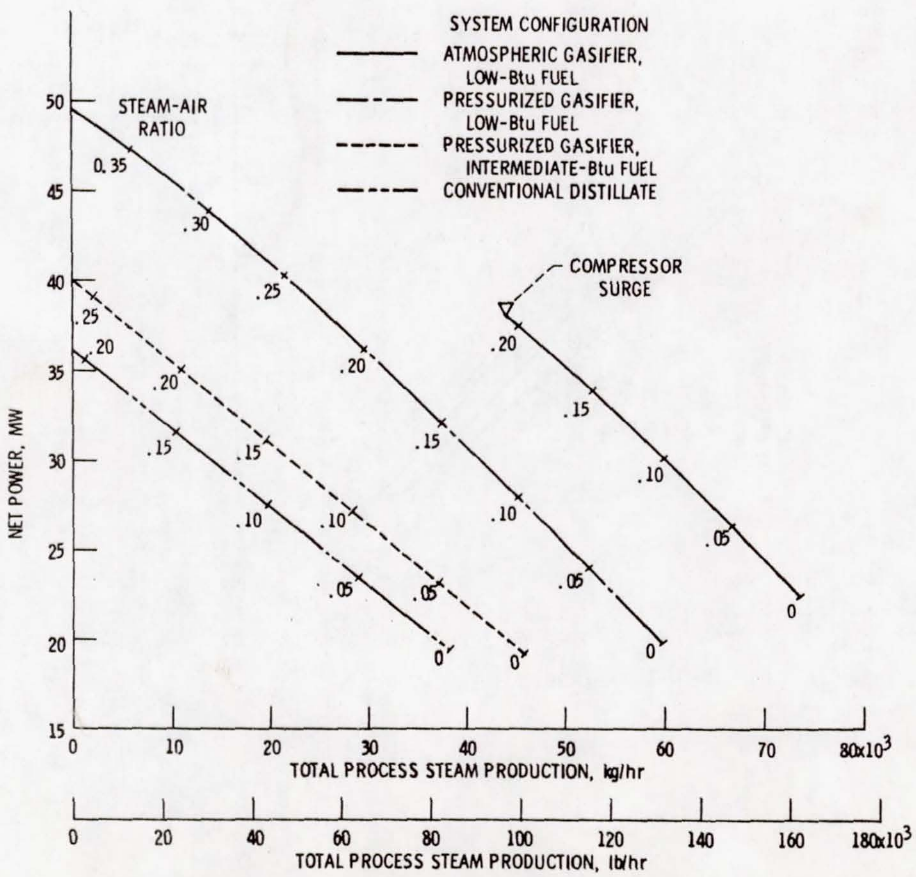

Figure 4. - Cogeneration system performance for fixed turbomachinery size of design option 1 for various fuels. Steam temperature, 482° C (900° F); steam pressure, 1.379 MPa (200 psia). (From ref. 9.)

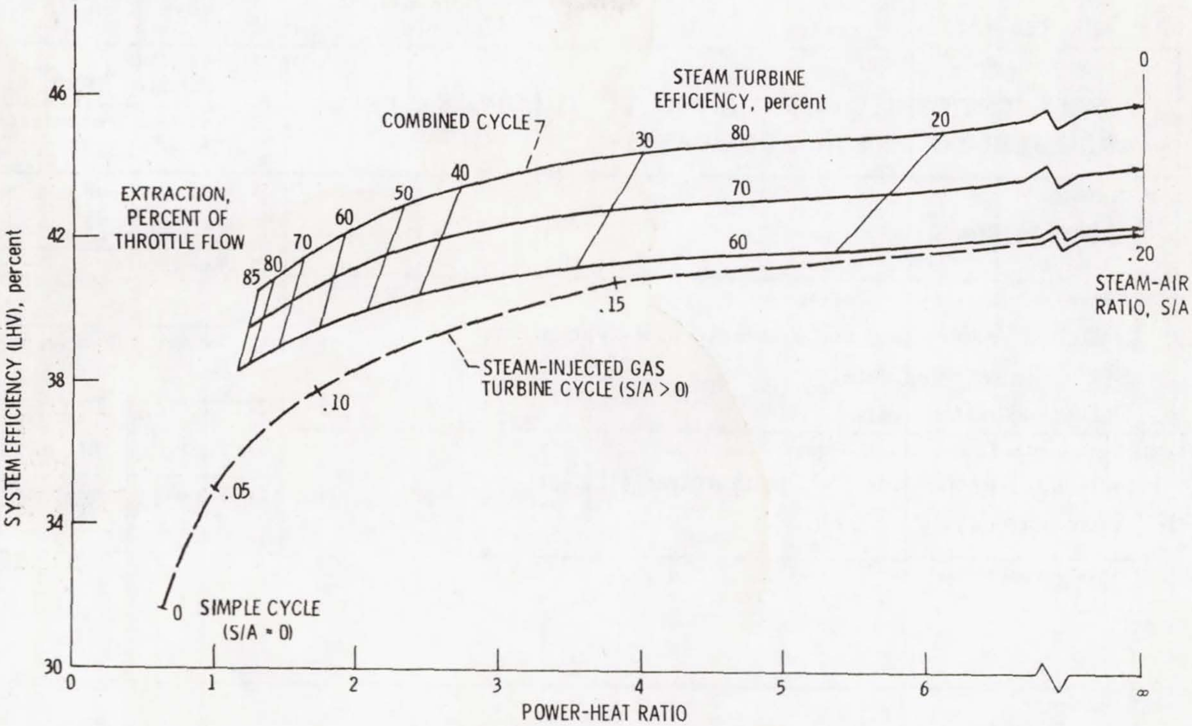

Figure 5. - Performance comparison over a range of power-heat ratios.

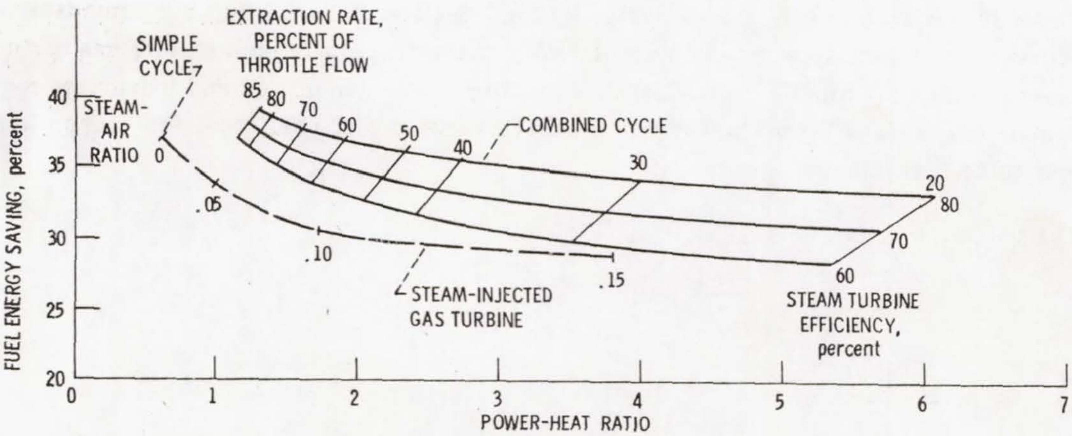

Figure 6. - Comparison of fuel energy savings (light distillate) over a range of power-heat ratios. Compressor pressure ratio, 11; turbine-inlet temperature, 2000° F; steam turbine temperature, 1000° F; steam turbine pressure, 1450 psia.

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16. Abstract

An assessment of the performance, economics, and potential application of steam-injected gas turbine cycles is presented. Results were arrived at by utilizing and expanding on information presented in the literature. The results are analyzed and compared with those for simple gas turbine and combined cycles for both utility power generation and industrial cogeneration applications. The efficiency and specific power of simple gas turbine cycles can be increased as much as 30 and 50 percent, respectively, by the injection of steam into the combustor. Steam-injected gas turbines appear to be economically competitive with both simple gas turbine and combined cycles for small, clean-fuel-fired utility power generation and industrial cogeneration applications. But for larger powerplants with integrated coal gasifiers, the economic advantages appear to be marginal.

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